Oil whip of a rotor supported in a poorly lubricated bearing

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Fluid whip is the post-instability-threshold, limit cycle, self-excited vibration of the rotor. It is a result of the fluid force generated due to rotor rotation in small radial clearances, such as in bearings. The fluid whip frequency is very close to the rigidly supported rotor natural frequency [1]. For the fully developed circumferential flow in the clearance, the fluid force contains a tangential component (a "cross-coupled stiffness" force) which has the forward direction (with direction of rotation). The resulting whip orbiting is always forward. A different situation occurs when the journal contact with the bearing is a dry sliding type. The friction-related tangential force points backward, and the resulting self-excited vibrations, if they occur, exhibit backward orbiting [2, 3]. An interesting situation occurs in the bearing when the lubricant supply pressure is low and causes partial starvation and mixed fluid/void lubrication. The resulting tangential force is then highly variable, from positive (forward) to potentially negative (backward), as the journal moves inside the bearing.

This article presents a case history of the rotor dynamic behavior when the lubricant starvation occurs in the rotor-supporting, fluid-lubricated bearing. In spite of basic mechanical system similarities, the rotor behavior here is different than the one presented in [4].

Experimental Setup

The experimental rotor rig consisted of a 1/10 hp motor (with a speed controller) driving, through a flexible coupling, a 3/8 inch diameter shaft carrying two 0.8 kg mass disks (Figure 1). The inboard bearing was a relatively rigid oilite bronze bushing. The outboard bearing was a cylindrical, fluid-lubricated bearing supplied with low (< 1 psi) pressure T10 oil. Bearing diametrical clearance was 13 mils. Due to gravity, the journal at rest was at the bottom of the bearing.

The rotor lateral motion was observed by two sets of proximity transducers in XY configuration at the outboard bearing and at rotor midspan, and a Keyphasor® transducer was used to provide speed and phase reference. The rotor startup data was captured by the DAIU 208 System and processed using ADRE® Software.

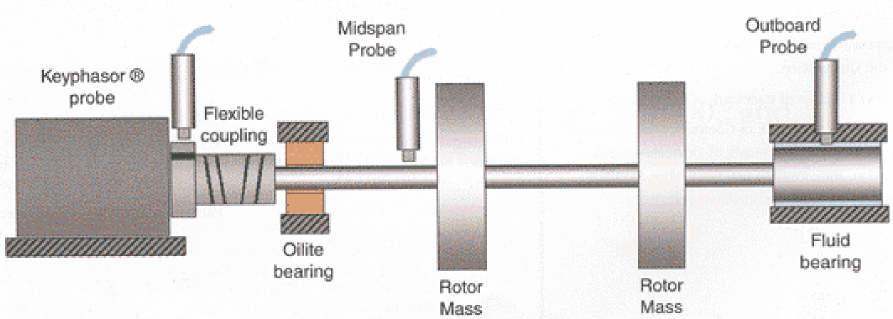

Figure 1 - Experimental rotor rig.

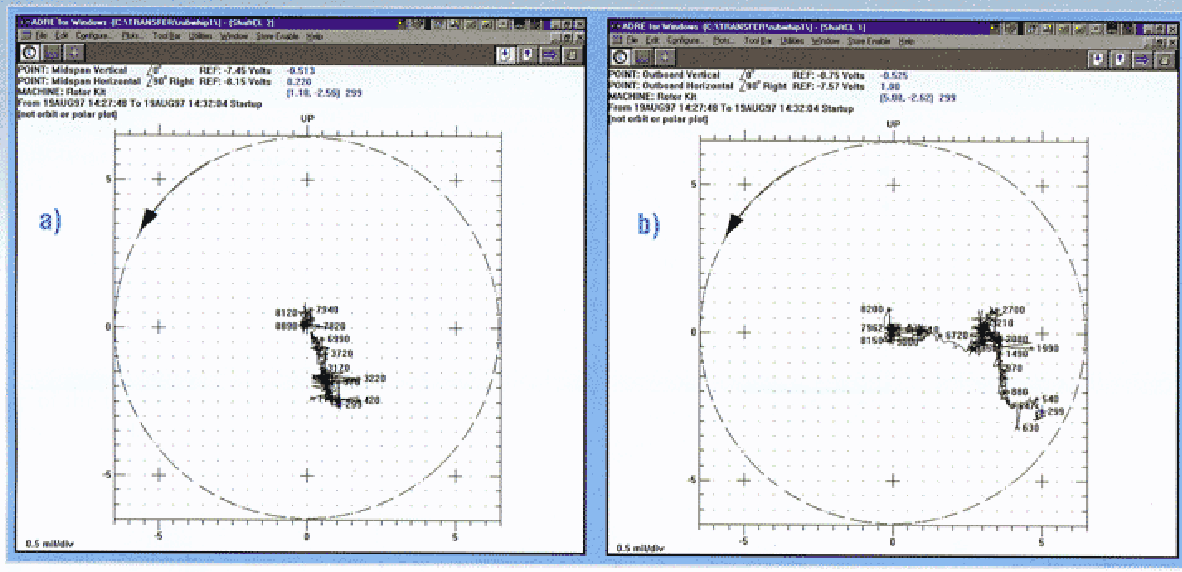

Figure 2 - Rotor midspan (a) and outboard bearing (b) shaft average centerline position during startup.

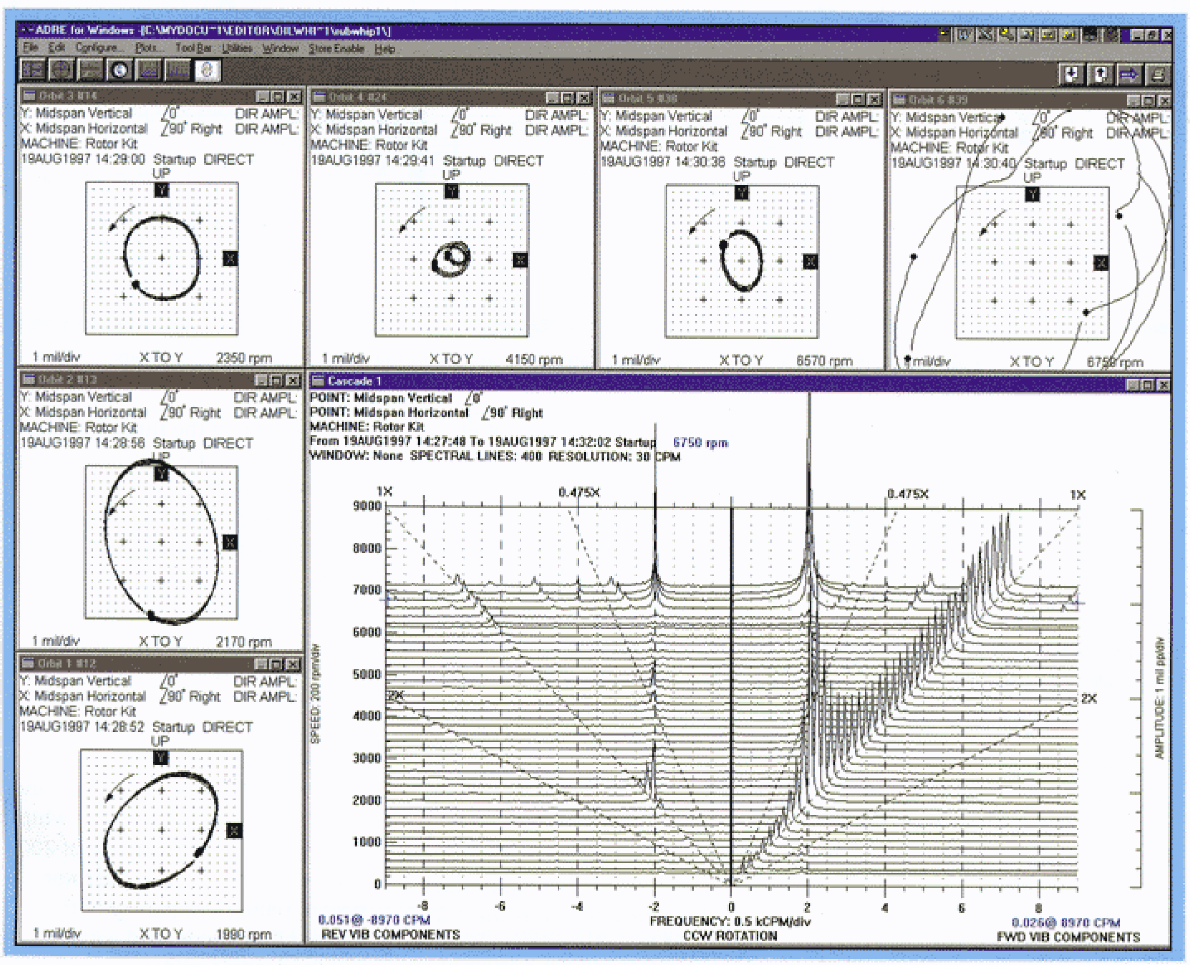

Figure 3 - Full spectrum cascade plot of the midspan lateral vibrations during startup, with selected orbits.

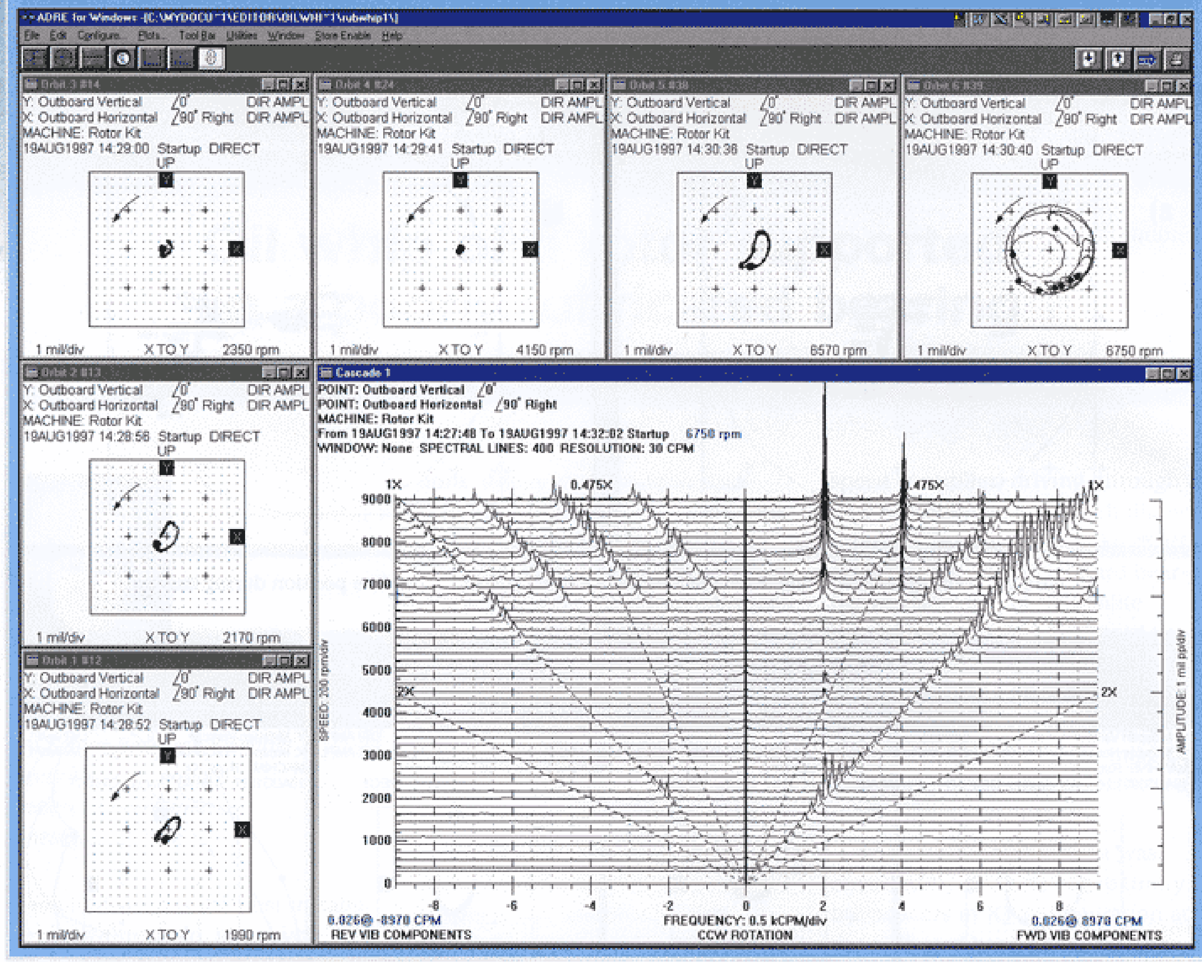

Figure 4 - Full spectrum cascade plot of the journal lateral vibrations during startup, with selected orbits.

Results of the experiment and discussion

The vibrational data taken during the rotor startup is presented in Figures 2 to 6. Figure 2 illustrates the rotor average centerline position at the midspan and outboard bearing locations during startup. The journal position at rest was at the bottom of bearing clearance. From zero to 300 rpm, the slow roll speed, the journal moves significantly to the right and up as the fluid wedge becomes active (Figure 2b). The journal continues moving up as the rotative speed increases up to approximately 6000 rpm. After the main instability threshold, which occurs at 6750 rpm, the journal centerline moves significantly to the bearing center, as high amplitude oil whip vibrations develop. The rotor midspan centerline (Figure 2a) moves much less than the journal centerline.

The full spectrum cascades (Figures 3 and 4) present the midspan and journal startup vibrations. The rotor is stable, exhibiting low-magnitude, elliptical 1X orbits until approximately 3800 rpm, when small, oil whip-type, self-excited vibrations occur (see rotor orbit at 4150 rpm, Figure 3). This speed is the rotor's natural frequency doubled (or rather, it is the $1/\lambda$ multiple of the natural frequency; λ = fluid circumferential average velocity ratio at the fluid-lubricated bearing). Since the journal is situated at relatively high eccentricity and the fluid pressure is low, at higher speeds the rotor stabilizes. There is not enough strength in the circumferential flow around the journal to sustain the whip vibrations. The banana-shape rotor orbit at 6570 rpm indicates that the journal centerline is at high eccentricity (Figure 4). The main onset of rotor instability occurs at 6750 rpm, and it leads to the self-excited oil whip vibrations with a frequency of 2000 cpm, corresponding to the rotor system natural frequency of the first bending mode, and with high amplitudes (see midspan and jour-

nal orbits at 6750 rpm, Figures 3 and 4). The journal orbit envelope is circular and nearly equal in magnitude to the bearing clearance; the midspan orbit is elliptical (due to an anisotropy of the rotor supports), and exceeds 40 mils pp. The rotor is dramatically vibrating at its first bending mode.

The full spectrum cascade plot confirms the ellipticity of the rotor mid-span whip orbit (31 mils pp of the forward component and 8 mils pp of the backward one), and the almost circular orbit of the journal (the reverse component is only approximately 0.5 mils pp, while the forward one is 9 mils pp).

The full spectrum cascade reveals also the nonlinearity of the physical phenomena involved in the rotor vibrations at speeds between 6750 and 9000 rpm. The presence of higher harmonics and sum/difference fractional vibration components are symptoms of the system nonlinearity. The relative magnitude of higher harmonics is larger in the journal spectra. The journal is the main location where the nonlinearities originate. The source of nonlinearity is the fluid film radial stiffness and damping. Some additional nonlinearity results from the geometric source of the highly bent rotor. At the journal, the amplitude of the forward component of the second harmonic of whip is approximately 1/3 that of the whip, while at the rotor midspan, the forward component of the second harmonic of whip is only approximately 1/60 of the whip forward amplitude. At high rotative speeds there are two major vibration components in the spectrum (1X due to rotor unbalance and the oil whip vibrations). Due to nonlinearity of the system, they create sideband sum/difference components. Note that the whip frequency, which was 2000 cpm at the instability threshold, slightly increases at higher rotative speed: to 2010 cpm at 7290 rpm and to 2030 cpm at 7740 rpm and higher speeds. This

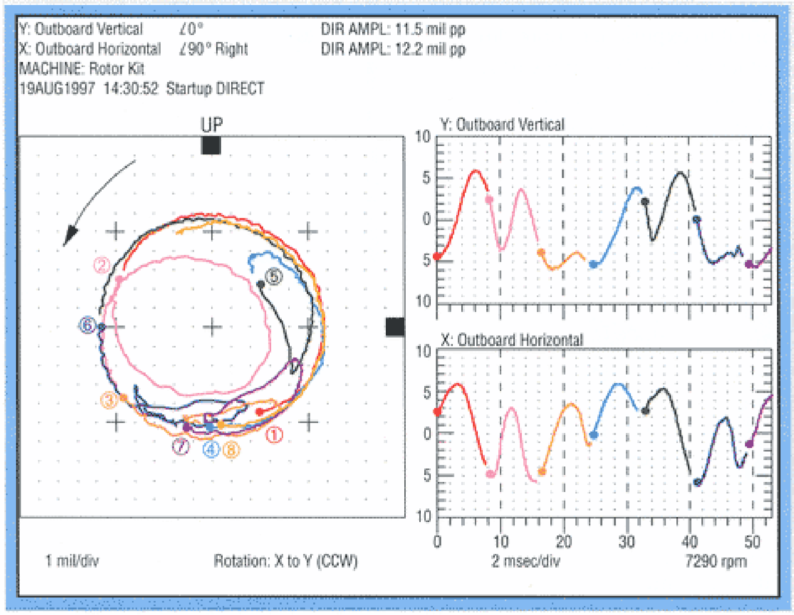

Figure 5 - Journal orbit and timebase waveforms at 7290 rpm.

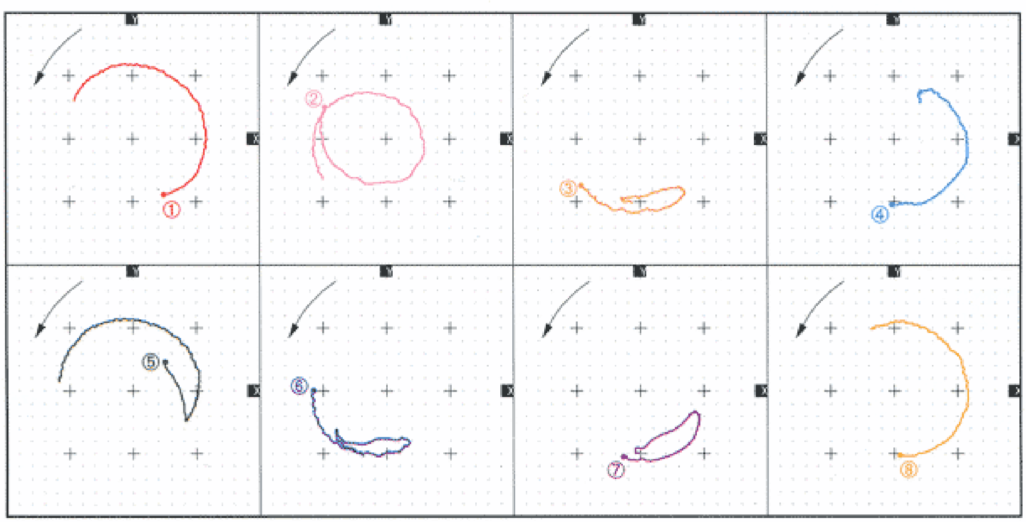

Figure 5-1. Orbits of the eight consecutive rotations of journal shown in Figure 5. Keyphasor dot marks start of orbit.

effect is best seen on the third harmonic of the whip (Figure 4).

The richness of the spectrum is due to the unusual behavior of the fluid film in the outboard bearing. The low-input pressure of the oil creates oil starvation and entrainment of air. Thus, the oil wedge produced by fully developed circumferential flow due to journal friction drag of the lubricant is no longer uniform. The voids allow the journal to experience more direct contact with the bearing surface. In this situation, which is similar to a rub, the flow-related forward-driving tangential force is occasionally replaced by a contact-friction-related backward driving force in the air voids. The tangential force becomes highly variable, from forward to possibly backward directions. When the journal/bearing contact occurs, the rotor forward precession may stop, then reverse for a short while until the pressure wedge sufficiently builds up, pushing the journal in the forward direction again. The result of this action is seen on the journal orbits (Figures 5 and 6). The numbers on the Keyphasor® dots indicate consecutive rotation periods of the rotor.

During the first rotation (Figure 5-1), the high amplitude vibrations are classic for the oil whip. During the next rotation, however, the journal makes an internal loop, as the whip amplitude decreases. During the third rotation, the journal reverses and makes a small loop. It continues forward for a while, then reverses again in three slightly larger loops, before reaching the eighth Keyphasor dot.

At the end of the fourth rotation, just before the fifth Keyphasor dot, the forward, oil-whip characteristic orbit reverses, and the journal makes a detour, for a short while going backward (approximately 1/4 of the rotation period), then recovers (the oil wedge succeeds in building up) and continues going forward.

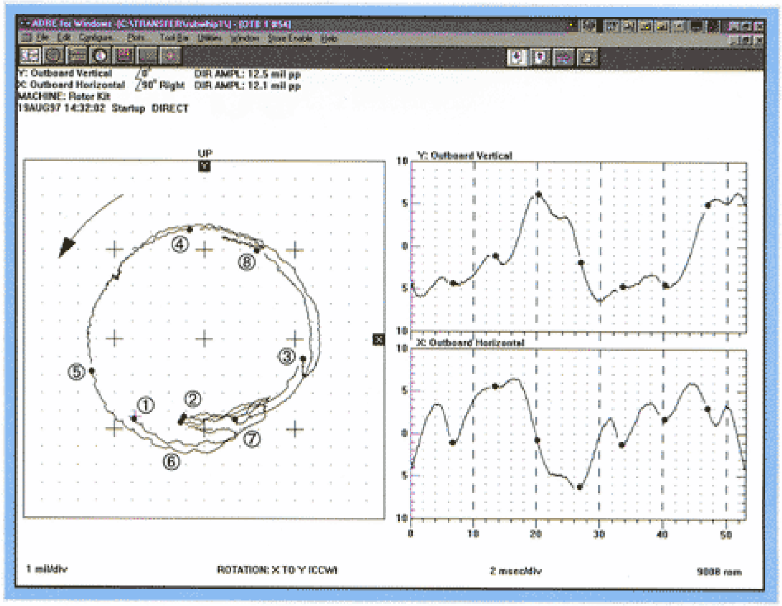

Figure 6 - Journal orbit and timebase waveforms at 9008 rpm.

The journal reversing for rotations 3, 6, and 7 occurs in the lower right quadrant, the same quadrant where the rotor centerline was at high eccentricity, thus in the area where the journal/bearing contact occurs. From the character of the orbital motion, it looks like when an oil void occurs, it causes the rotor forward precessional motion to stop and reverse by giving it a reverse impulse. At that moment, however, the journal moves away from the bearing surface, so that oil film has a chance to build up again. This scenario repeats four times within 5 rotations (Figure 5).

At high speed, the journal orbit is much smoother; several smaller backward loops still appear (Figure 6). The higher speed provides a larger energy into the forward motion, thus the backward impulses from the surface contacts are relatively smaller [5].

Final remarks

The case history described above provides an impressive, qualitative scenario on physical phenomena occurring in the fluid film of a poorly lubricated bearing. The rotor vibration signature is a reflection of the physical phenomena taking place at the bearing. By using the appropriate vibration data processing, these phenomena can be unveiled, explained, and eventually mathematically modeled [5].

References

1. Muszynska, A., Bently, D. E., Fluid-Induced Instabilities of Rotors: Whirl and Whip – Summary of Results, Orbit, BNC, March 1996, pp. 7-15.
2. Muszynska, A., Effects of an Oversize, Poorly Lubricated Bearing on Rotordynamic Response. Part I: Experimental Results, Part II: Analytical Modeling, Rotating Machinery 1992, Vol. 2, Ed. Wen-Jei Yang & J. H. Kim, Begell House, Inc., New York, Walingford, U.K., 1992, pp. 1105-1125.
3. Muszynska, A., Rotor/Seal Full Annular Rub, Bently Rotor Dynamics Research Corporation, Report No. 2/84, Proc. of the Senior Engineering Seminar, Carson City, Nevada, June 1984.
4. Muszynska, A., Petchenev, A., Bently, D. E., Vibration Response of a Rotor Supported in One Rigid and One Poorly Lubricated Fluid Film Bearing, Orbit, BNC, v. 17, No. 3, 1996, pp. 7-9.
5. Muszynska, A., Yuan, J., Simulation Results for the Oil Whip of a Rotor Supported in a Poorly Lubricated Bearing, BRDRC Report No 3, 1998.